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(54) Vibration damping handle

(57) A vibration-damping handle for a hand-operated tool such as an electro-mechanical motive grinder, or for a control member of a vehicle which is subjected to vibration, said handle being connected to a source in the form of a vibration body (10) via a tie-rod (11) that extends from the centre of connection means on said vibration body, and having a first vibration-damping body (12) connected to the tie-rod at a position close to the connection means and a second vibration-damping body (14) attached at the free end of the tie-rod or of a further tie-rod (13) coaxial with the tie-rod (11), and a hand-grip (15) secured to the first vibration-damping body (12) and extending to cover the second vibration-damping body (14) to bring the centre of the hand-grip into agreement with a vibration node where the amplitude of vibration transmitted from the vibration body is minimal.

FIG. 6

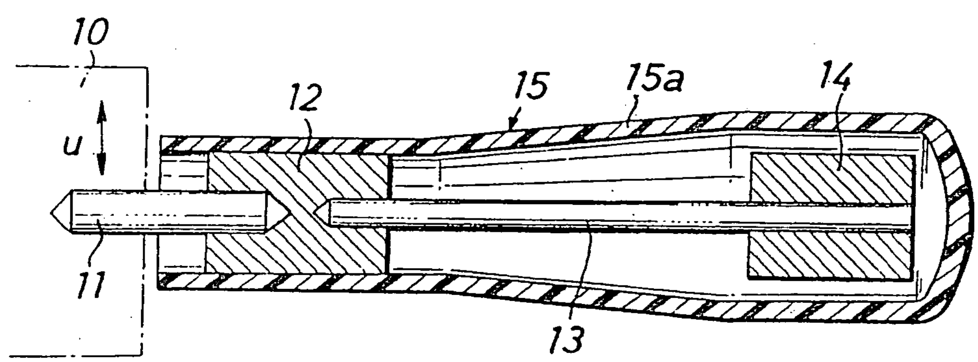

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FIG.1

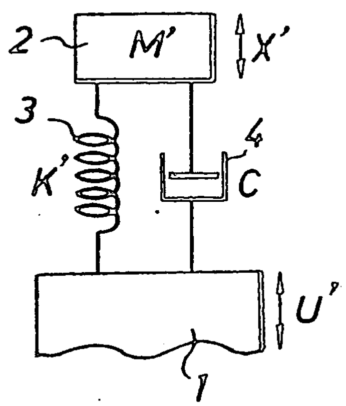


FIG.2

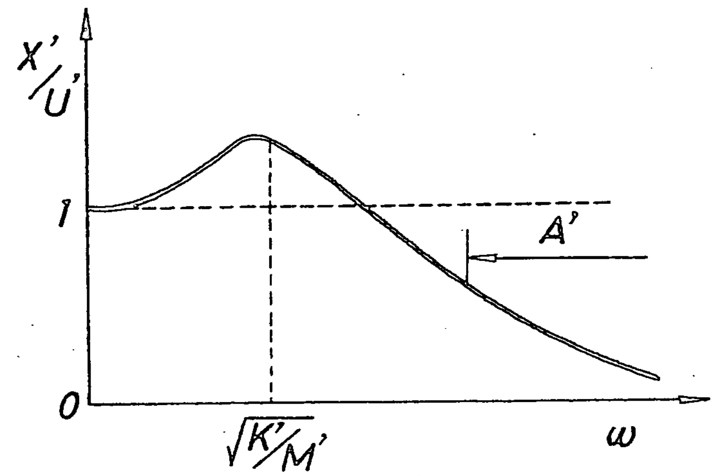


FIG.3

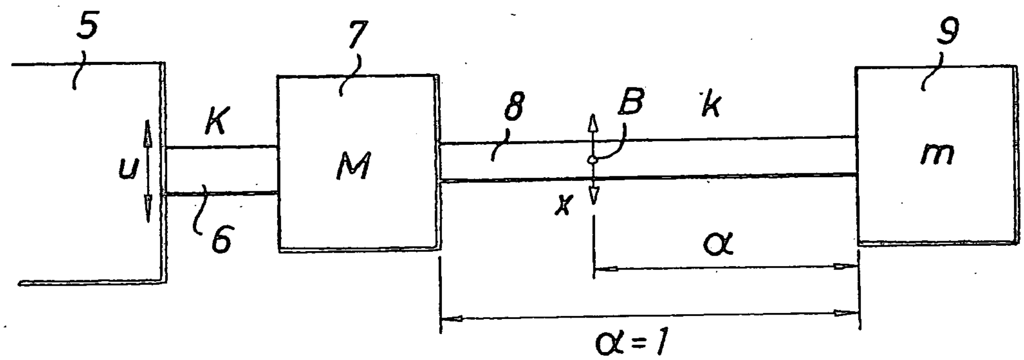


FIG. 4

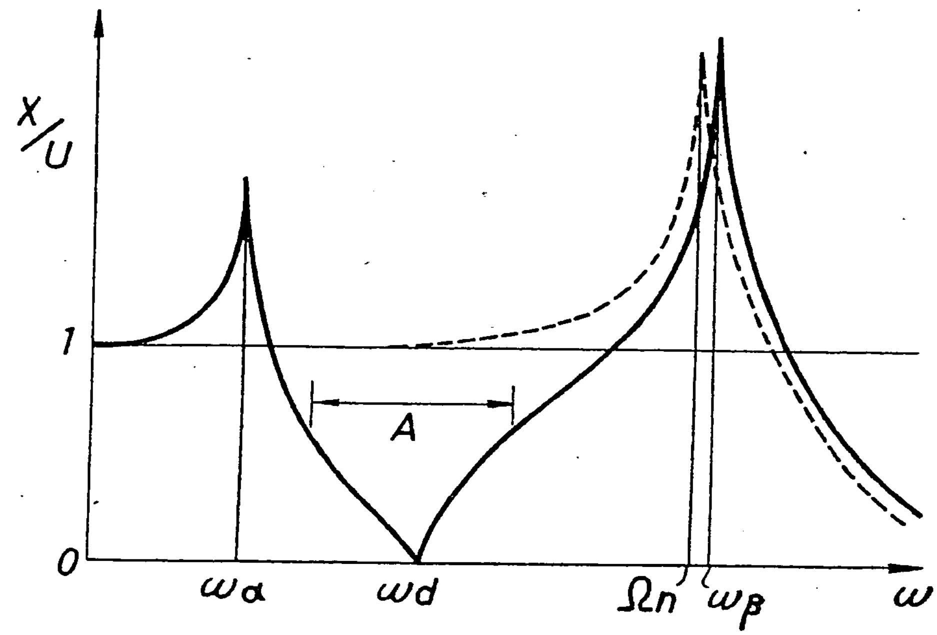
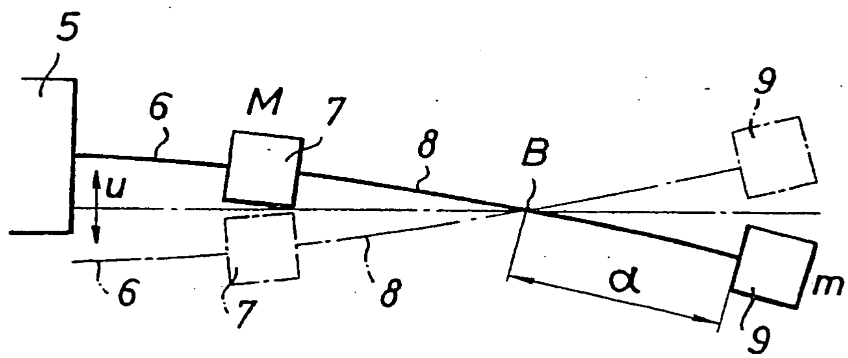


FIG. 5



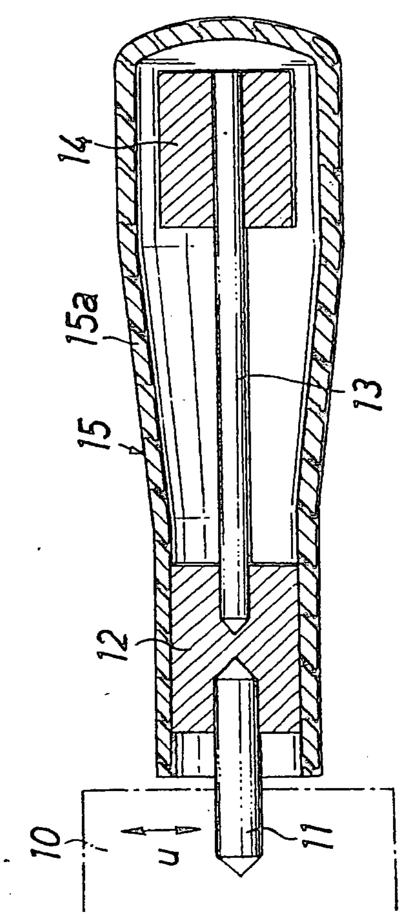

14
15a
15
12
10
u
11
13

FIG. 6

FIG. 7

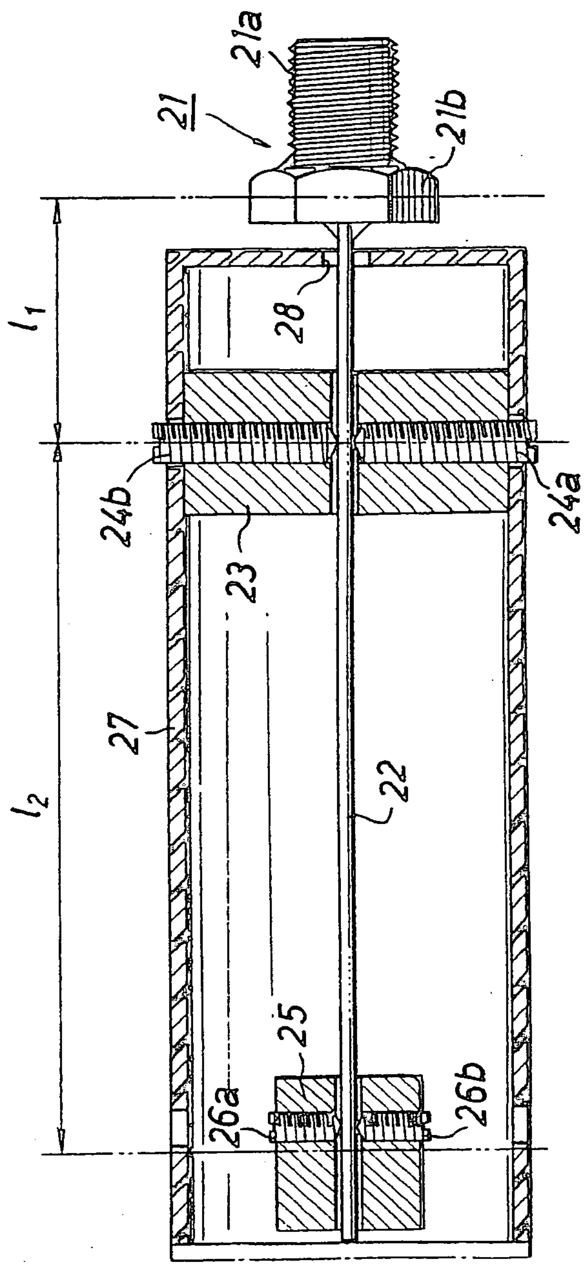

21
21a
21b
28
l_1
l_2
24b
24a
23
27
22
25
26a
26b

FIG. 8

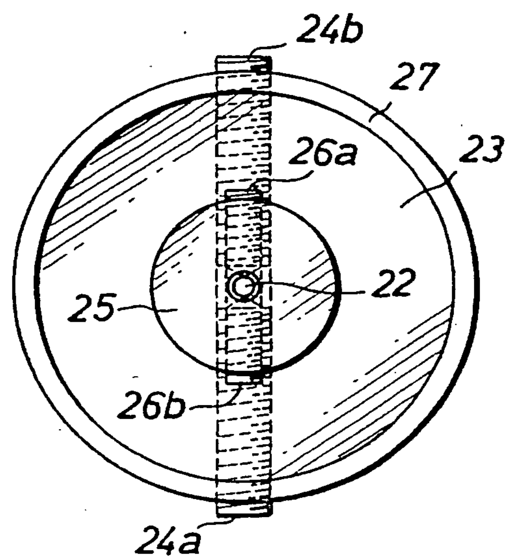


FIG. 9

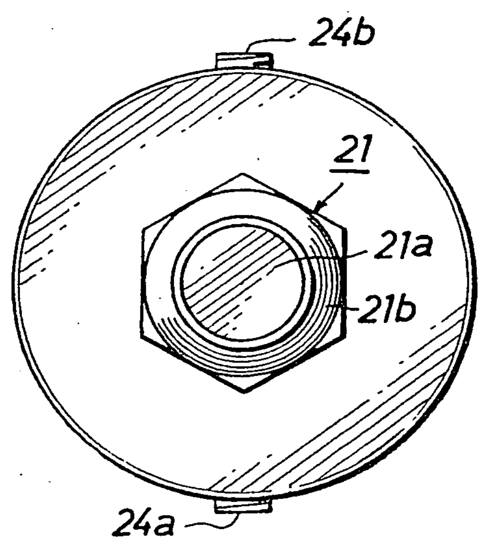


FIG. 10a

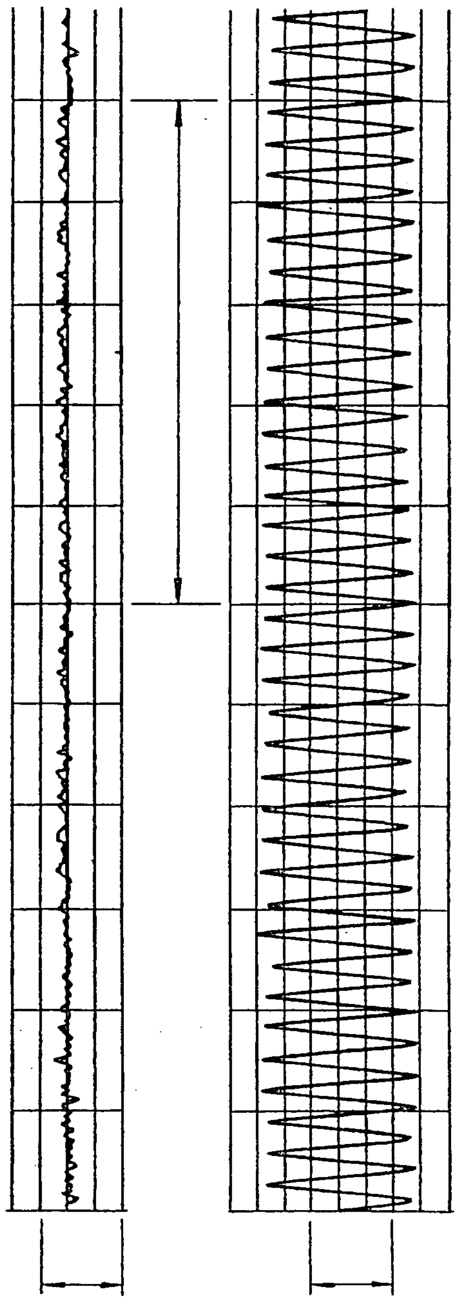
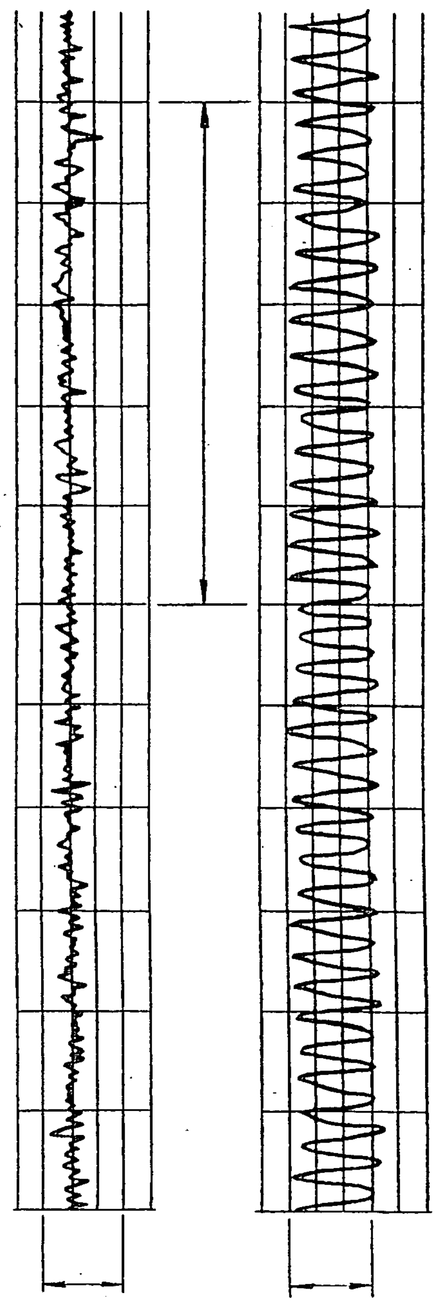


FIG.10b



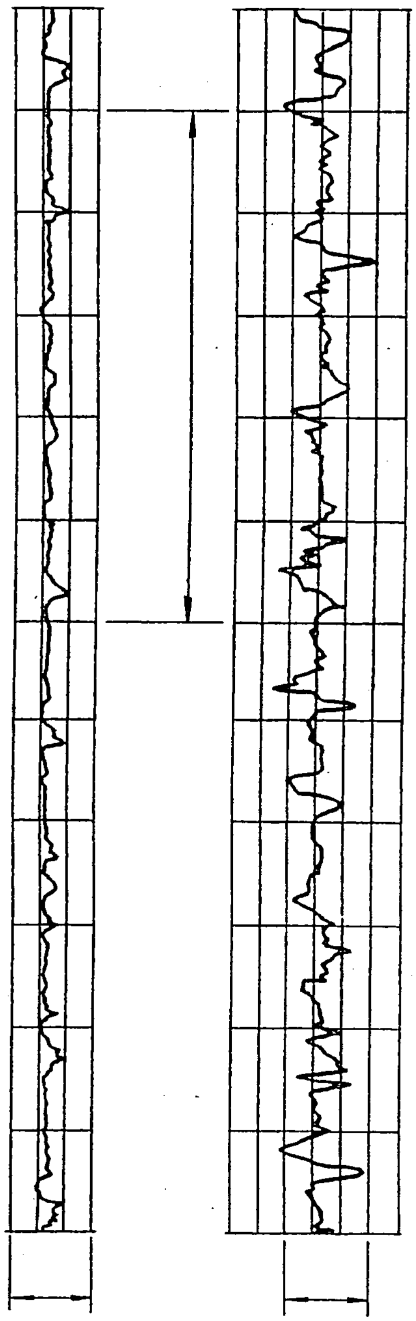


FIG. 11a

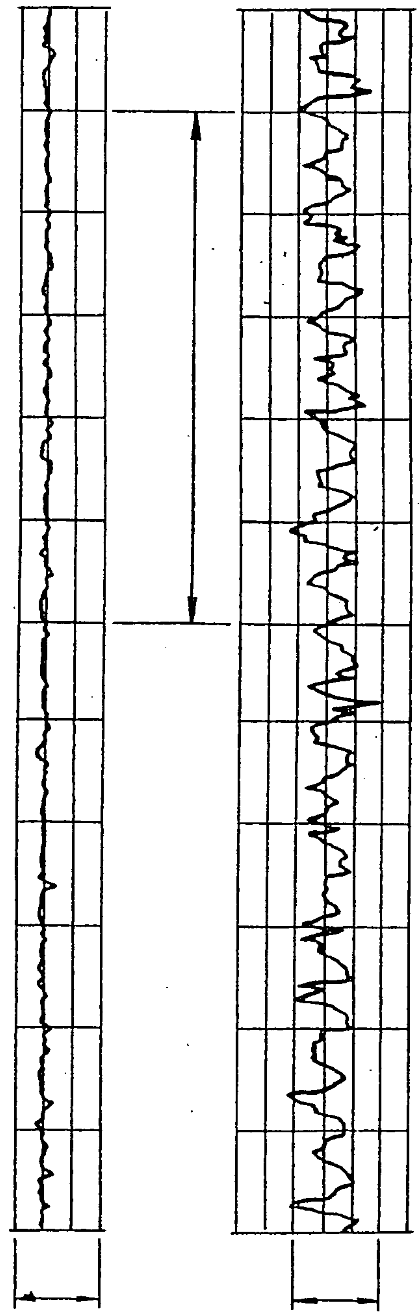


FIG. 11b

SPECIFICATION
Vibration damping handle

The present invention relates to vibration-damping handles for use with a vibration source such as a hand-operated pneumatic or electro-mechanical tool, e.g. a grinder, drill, a chain saw or like machinery or vehicles containing a vibrating member liable to pass vibration to an operator's control handle which is capable of substantially isolating the transmission of the vibrations to its hand-grip, and which is capable of improving the operational control effectiveness of the handle.

For the prevention of excessive vibration of the handles of hand-operated tools or machines which themselves generate vibrations, a vibration preventive rubber member is frequently provided between the vibration source and the handle to isolate the vibrations.

However, when such vibration preventive rubber member is used, a force proportional to the velocity of the vibration is transmitted to the handle, and it is difficult to sufficiently isolate vibrations which may cause a health hazard, such as a white finger disease. Further, with use of an elastic material such as a vibration preventive rubber, the effective control achievable by handle operation becomes so flexible that a proper management of the tool or machine cannot be ensured, and it is likely to lead to wobble or stagger, which can be a danger in operation.

One object of the present invention is to provide a construction which substantially overcomes the above-mentioned difficulties inherent to the conventional device, by means of a handle that is substantially rigidly connected to a vibration source, but is capable of effectively isolating hazardous vibrations.

When a vibration system comprising springs and weights and having multiple degrees of freedom, is vibrated, there exists one or more vibration nodes where the vibration amplitude is zero or minimal, and by designing a vibration isolation system so that the centre of a hand-grip coincides with a vibration node the vibration source could be effectively isolated.

The present invention consists in a vibration damping handle for use with a machine or a power tool which, when operating, acts as a source of vibrations, said handle comprising a connection member for rigid connection to said machine or tool, a tie-rod projecting from said connection member to a first vibration-damping body relatively close to said connection member, said tie-rod or a further tie-rod co-axial with said tie-rod extending to support a second vibration-damping member at or adjacent its free end, and a hand-grip secured to the first vibration-damping member and extending concentrically with respect to said or each said tie-rods to cover said second vibration damping body.

The invention will now be described with reference to the drawings, in which:—

Figure 1 is a schematic theoretical diagram illustrating a vibration model demonstrating the characteristics of a conventional method of vibration-damping;

Figure 2 is an explanatory graph showing the frequency characteristics of the amplitude ratio between the amplitude of a vibration source and the amplitude of a handle for the vibration model shown in Figure 1;

Figure 3 is a schematic theoretical diagram illustrating a vibration system of a model constructed in accordance with the present invention;

Figure 4 is an explanatory graph showing the frequency characteristics of the amplitude ratio between the amplitude of the vibration source and the amplitude of a point on a spring in the vibration system shown in Figure 3;

Figure 5 is an explanatory schematic diagram illustrating an example of the manner of vibration at $\omega = \omega_d$ in the vibration system shown in Figure 3;

Figure 6 is a schematic sectional view illustrating one exemplary embodiment constructed in accordance with the present invention;

Figure 7 is a schematic cross-sectional side view of another exemplary embodiment constructed in accordance with the present invention;

Figure 8 is an end view of the handle shown in Figure 7 as viewed from the left;

Figure 9 is an end view of the handle shown in Figure 7 as viewed from the right; and

Figures 10 and 11 are two pairs of explanatory comparative graphs showing the acceleration of the vibrations in cases for a handle constructed in accordance with the present invention and for a conventional handle device, each attached to two types of commercially available hand-operated electro-mechanical grinders.

As the present invention is based on a novel principle which is quite different from the principle used in conventional vibration preventive handles, the underlying principle of the present invention will first be explained and compared with the conventional vibration preventive theory, before describing the exemplary embodiments of the present invention.

Figure 1 illustrates the principle used in a conventional method of vibration prevention. A vibration body 1 is part of a tool or machine assumed to vibrate with an amplitude U' and a frequency of ω. A spring 3 having a spring constant K' is provided in parallel with a damper 4 having a damping coefficient C to couple the vibration body 1 and a handle 2 having a mass M'. The spring 3 and the damper 4 are theoretical components of a vibration damping rubber member.

When the vibration body 1 vibrates, vibration is transmitted via the spring 3 and the damper 4 to

the handle 2, whereupon the handle 2 is vibrated. If the amplitude of the handle vibration is X′, the frequency-dependent characteristics of the amplitude ratio X′/U′ are approximately as shown by the curve in Figure 2. Where the ratio X′/U′ is smaller than unity, the vibration of the vibration body 1 transmitted to the handle 2 has been damped. Accordingly, the frequency range within which this vibration preventive method becomes effective is limited to the area indicated by an arrow A′ in figure 2, which commences at a frequency considerably higher than the natural frequency $\sqrt{K'/M'}$ of the handle 2. In other words, in order to obtain a maximum effectiveness by this vibration preventive method, it is necessary that the natural vibration frequency $\sqrt{K'/M'}$ be set at a relatively low level. In order to set the value $\sqrt{K'/M'}$ at a lower level, it is necessary either to reduce K′ or to increase M′. If the value K′ is reduced, the rigidity of the handle 2 with respect to the vibration body 1 is lessened. On the other hand, if the value M′ is increased, the weight of the handle may become excessively heavy.

Thus, according to the above mentioned conventional vibration preventive principle, any enlargement of the effective vibration preventive frequency range inevitably involves no reduction of the weight of the handle or increased operational control of the handle. It is apparent that with this vibration preventive method, it is essentially impossible to obtain a handle having a high vibration preventive effectiveness, a superior operational control and a light weight.

The novel principle of the vibration preventive method used in any embodiment constructed in accordance with the present invention will be explained with reference to figures 3, 4 and 5.

In figure 3, the theoretical schematic diagram of a vibration system equivalent to a handle constructed in accordance with the present invention is shown. This vibration system comprises a vibration body 5 whose vibrations *u* are in the direction indicated by an arrow in the figure (i.e. an up-and-down direction so drawn), a spring coupling 6 extending in a direction at a right angle to the vibration direction (i.e. in a horizontal direction as drawn), a weight 7 attached to the remote end of the spring coupling 6, a spring coupling 8 extending co-axially with the spring coupling 6, and a weight 9 attached to the remote end of the spring 8. The spring constants of the spring couplings 6 and 8 in the up-and-down direction, are K and *k* respectively. The masses of the weights 7 and 9 are M and *m* respectively. (Strictly speaking, there exists a bending moment acting on each spring and each weight. However, for the convenience of explanation of the principle, this moment may be ignored).

If the vibration phase *u* of the vibration body 5 is set at $u = U\sin\omega t$, and assuming a vibration phase *x* at a selected point B on the spring coupling 8 is set at $x = X\sin(\omega t + \psi)$, (where ψ is a phase angle), the resultant amplitude ratio X/U is represented by the following formula, as a function of the frequencies:

$$\frac{X}{U}(\omega) = \frac{\left(1 - \alpha\frac{m}{k}\omega^2\right)}{\frac{M}{K}\frac{m}{k}\omega^4 - \left(\frac{M}{K} + \frac{m}{K} + \frac{m}{k}\right)\omega^2 + 1}$$

$$= \frac{\alpha\frac{K}{M}(\omega_d^2 - \omega^2)}{(\omega_\alpha^2 - \omega^2)(\beta^2 - \omega^2)} \quad \cdots\cdots\cdots\cdots (1)$$

where,

$$\omega_d = \omega_n\sqrt{\alpha}$$

$$\omega_\alpha^2 = \frac{1}{2}\left\{(\omega_n^2 + \mu\omega_n^2 + \Omega_n^2) - \sqrt{(\omega_n^2 + \mu\omega_n^2 + \Omega_n^2)^2 - 4\Omega_n^2\omega_n^2}\right\}$$

$$\omega_\alpha^2 = \frac{1}{2}\left\{(\omega_n^2 + \mu\omega_n^2 + \Omega_n^2) + \sqrt{(\omega_n^2 + \mu\omega_n^2 + \Omega_n^2)^2 - 4\Omega_n^2\omega_n^2}\right\}$$

Further, α is a distance from the weight 9, as standardised with regard to the distance between the weight 7 and the weight 9. The range of α is $0<\alpha<1$.

In the above formula, Ω_n and ω_n represent the natural frequencies of the weights 7 and 9, respectively, μ represents a mass ratio between the weights 7 and 9, so that $\Omega_n = \sqrt{K/M}$, $\omega_n = \sqrt{k/m}$, $\mu = m/M$.

As is apparent from the formula (1), if the expression

$$\omega_d (= \omega_n/\sqrt{\alpha} = \frac{1}{\sqrt{\alpha}} \cdot \frac{\sqrt{m}}{k})$$

is brought to agree with the vibration frequency ω of the vibration body 5, the numerator of the formula (1) becomes zero, whereby it is possible to bring the amplitude X to zero at one point on the spring 8, i.e. point B in figure 3) and a vibrational node will be formed at point B. In figure 5, the manner of vibration of this vibration system is schematically illustrated for a case where $\omega = \omega_d$. The frequency characteristics of the formula (1) become generally as shown by the solid line curve in the graph shown in figure 4. It will be seen that over a wide frequency range indicated by an arrow A with ω_d approximately at its centre the value X/U is less than unity. In embodiments of the present invention, the centre of the hand-grip of the handle is set at the vibration node.

On the other hand, in the conventional dynamic vibration absorption device, the weight 7 corresponds to the handle ($\alpha = 1$ in the formula (1)), and the spring couplings 8 and 9 representing a dynamic vibration absorption device, are complementarily attached to the weight 7, whereby it is intended to isolate the weight 7 from the vibration of the vibration body 5 by approximately selecting various values to satisfy $\Omega_n \doteqdot \omega_n$ and $\mu \ll 1$. However, in this case, there is only a narrow frequency range in which $X/U < 1$ is satisfied, and if the vibration frequency ω of the vibration body 5 shifts slightly from ω_n, the vibration preventive effect is greatly reduced. Further, if $\mu = m/M$ is allowed to take a greater value (namely, if m is allowed to be greater), the handle becomes heavier.

The present invention has improved the vibration preventive system beyond the conventional concept for the dynamic vibration absorption device, and the interesting features of the invention resides in the selection for $0 < \alpha < 1$, $\Omega_n > \omega_n$, $\mu = 1$, namely for $M \doteqdot m$, $K > k$. By this selection, the change of X/U in the vicinity of ω_d becomes moderate, and besides, the range represented by the arrow A becomes wide. Accordingly, even when there is a departure in the vibration frequency ω of the vibration body 5 from ω_d to a substantial extent, it is still possible to isolate the vibration. Further, as M = m, the weight of the handle may be reduced, and as K may take a greater value, the weight 7 can be rigidly connected to the vibration body 5.

Two preferred exemplary embodiments of the present invention prepared on the basis of the above-mentioned principle will now be described with reference to figures 6 to 9 of the accompanying drawings, figure 6 showing a first exemplary embodiment, and figures 7 to 9 showing a second exemplary embodiment.

Figure 6 shows a schematic representation of a vibration source 10, such as a casing of a vibrating tool or machine, which vibrates in an up-and-down direction, as drawn. A first cantilever type spring member in the form of a tie-rod 11 has one end embedded and welded in the vibration source 10 to ensure a secure connection, and the free end extending horizontally, as drawn, to carry a first vibration-damping body 12 of cylindrical shape. A second cantilever type spring member in the form of a further tie-rod 13 extends co-axially with respect to the tie-rod 11 from the body 12 to a second vibration-damping body 14 of cylindrical shape attached to the front end of the second cantilever type spring member 13. The vibration-damping body 14 is slidable on the spring member 13, and means (not shown) are provided for locking it in its adjusted position. A hollow closed hand-grip member 15 of cylindrical shape is provided to extend from the first body 12 to cover the second spring member 12 and the second body 14, with adequate clearance therefrom.

The term "vibration-damping body" is intended to refer to any member of the assembly whose mass is of such a magnitude that the mass itself significantly affects the frequency-dependent vibrational characteristics of the assembly.

When the source 10 vibrates, vibrations will be transmitted via the first cantilever type spring member 11 to the first body 12, whereupon this body 12 will be vibrated. The vibrations of the first body 12 will be transmitted via the second cantilever type spring member 13 to vibrate the second body 14. The vibration of the first body 12 will also be transmitted to the hand-grip 15.

In this embodiment, the spring constant of the first cantilever type spring member 11 in an up-and-down direction (as drawn) is set to be substantially larger than that of the second cantilever type spring member 13, and accordingly, the first body 12 can be deemed to be substantially rigidly connected to the vibration source 10. Further, the mass of the first body 12 is set to be substantially equal to the mass of the second body 14, and the mass of the hand-grip 15 is much lighter than that of either body, 12 or 14. When this embodiment is compared with the vibration system of figure 3, it is seen that the cantilever type spring members 11 and 13 correspond to the spring couplings 6 and 8 respectively, and the bodies 12 and 14 substantially correspond to the weights 7 and 9, respectively.

Accordingly, a vibration node corresponding to point B in figure 3 is formed at a certain point on the second cantilever type spring member 13, and the centre 15*a* of the hand-grip 15 is approximately positioned on this vibration node. As is apparent from the above description of the principle, the vibration from the vibration source 10 will be substantially isolated from the centre 15*a* of the hand-grip 15.

Further, the masses of the first and second vibration-damping bodies 12 and 14 may be substantially equivalent, and accordingly, the hand-grip and its assembled components may be constructed to be of low weight. Furthermore, the location of the vibration node can be adjusted by sliding the second body 14 along the second cantilever type spring member 13.

In the above-described embodiment, the shape of the hand-grip 15 is only required to be distanced so that it will not be in contact with the second body 14, even when the latter is vibrated, and the hand-grip 15 may be provided with openings at appropriate locations. The fastening means for securing the first cantilever type spring member 11 and the vibration source 10 or to the first damping body 12 is not limited to welding, but may employ other means, such as screws. Furthermore, the second cantilever type spring member 13 and the hand-grip 15 may appropriately be non-linear in a horizontal plane relative to the vibration source 10.

The alternative exemplary embodiment of the present invention shown in figure 7 comprises a connection member 21 in the form of a bolt 21*a* secured to a nut 21*b*, being welded together after assembly. The connection member may be formed as an integral unit, for instance, by milling, and can be of any required form, as determined by the intended field of application.

Female screw threads may be omitted, and replaced by rivetting, welding or any other suitable fastening means, so long as it is capable of establishing a secure connection with the vibration source such as a casting or housing (not shown) of a electro-mechanical or a pneumatic tool, or a machine assembly such as a motorcycle. In all cases, the assembly will require locked or shakeproof fixing of all individual elements, in conventional manner, and the individual means have not been illustrated for the sake of clarity.

At the centre of the face of the connection member there is securely attached by welding, screwing, or other suitable fastening means, a tie-rod 22, preferably a thick piano wire, which extends togwards the left, and a first vibration-damping body 23 is secured on the tie-rod at an intermediate point. In the illustrated embodiment, two screws, 24*a* and 24*b*, are used as the securing means. However, other securing means, such as welding or other types and arrangements of screws may be used.

A second vibration-damping body 25 is secured at or adjacent to the free end of the tie-rod 22. In the illustrated embodiment, screws 26*a* and 26*b* are used as the securing means. However, other suitable securing means may be used, as mentioned above, and it is not essential in all cases to provide for adjustment of the position.

The first body 23 is spaced from the base of the connection member 21 by a distance l_1, which is from about 50 to about 70 mm in the case of a hand-grip of a motorcycle handle-bar, and the first body 23 is spaced from the second body 25 by a distance l_2 which is from about 70 to 120 mm. Thus l_1 is always shorter than l_2. The diameter of the tie-rod 22 is from about 5.2 to about 8.0 mm. The tie-rod may have a cross-section other than circular, for example it may be square, hexagonal or flat in cross-section.

A hand-grip 27 made of a semi-rigid material such as a synthetic resin, rubber, hard paper or wood, is formed in a generally circular sleeve or tubular shape, and is secured to the body 23, with a length sufficient to cover the second body 25. Referring to figure 7, the left-hand end of the hand-grip 27 may be open as illustrated, or may be closed by a cap indicated by one additional broken line. As illustrated in figure 7, the right-hand end of the hand-grip 27 may be closed except for clearance opening 28 through which the rod member 22 passes.

This second embodiment acts to provide a vibration preventive effect in the same manner as that described for the first embodiment, with special reference to figure 4, whereby no hazardous vibrations will be transmitted to the hand-grip 27. In this embodiment, the locations of the first body 23 and the second body 25 are adjustable on the tie-rod 22, and thus, it is possible to obtain optimal vibration isolation by adjusting their positions. Figure 8 shows an end view of an open-ended hand-grip, and figure 9 a view from the other end. A respective adjusting hole is provided in the hand-grip to give access to each fixing screw, and if three or more symmetrically disposed screws are used to ensure positive centering, the access holes will be appropriately repositioned.

The results obtained by a vibration test carried out with use of two types of commercially available hand-operated electro-mechanical grinders equipped with its standard handle and then a handle of the type shown in figure 6 will now be explained with reference to the graphs shown in figures 10 and 11. In the experimental handle prepared for trial, a piano wire having a diameter of 6 mm (the effective length being 10 mm) was used for the first cantilever type spring member 11 and a piano wire having a diameter of 4.5 mm (the effective length being 60 mm) was used for the second cantilever type spring member 13. The masses of the first and second bodies 12 and 14 were each adjusted to be 100 g.

The graphs shown in figure 10 relate to a hand-operated electro-mechanical grinder marketed under the trade name of Toshiba as Model DG—125B, and the graphs of figure 11 relate to a grinder marketed under the trade name of Mitake as Model AG—180. These hand-operated electro-mechanical grinders were first equipped with their standard handles, and then a handle of the design shown in figure 6. The upper graph of figure 10*a* shows the acceleration of left-to-right vibrations transmitted to the handle constructed in accordance with the present invention, whilst the corresponding vibrations with the standard handle are shown below, for comparison. The measurements were carried out while

idling the electric grinders, and the measurement was made at a point 10 cm inward from the front end of the handle. In figure 10*b* the upper graph shows the acceleration vibrations in the up-and-down direction of the handles constructed in accordance with the invention, and the result with the standard handle are shown below for comparison. The upper and lower graphs of figures 11*a* and 11*b* are arranged in the same manner, for the other make of grinder. In each graph the horizontal axis represents time, and the vertical axis represents acceleration. To the left of each graph an arrow indicates the amplitude of acceleration due to gravity, for use as a reference value, and the horizontal arrows indicate unit time. In each case for the handle constructed in accordance with the invention, a remarkable vibration-damping effect is observed and it can be seen that the acceleration amplitudes in both the up-and-down direction and the left-and-right direction are reduced to about 1/5 by a handle constructed in accordance with the present invention.

Our copending United Kingdom Patent Application No. 8119221 (V734); No. 8119223 (V735); and No. 8119224 (V751); all of even date, relate to other forms of vibration-damping handles.

CLAIMS

1. A vibration-damping handle for use with a machine or a power tool which, when operating, acts as a source of vibrations, said handle comprising a connection member for rigid connection to said machine or tool, a tie-rod projecting from said connection member to a first vibration damping body relatively close to said connection member, said tie-rod or a further tie-rod co-axial with said tie-rod extending to support a second vibration damping member at or adjacent its free end, and a hand-grip secured to the first vibration damping member and extending concentrically with respect to said or each said tie-rods to cover said second vibration damping body.

2. A handle as claimed in claim 1, in which there are two said tie-rods, that tie-rod carrying said first vibration-damping body having a spring constant in the transverse vibration direction that is greater than that of the second tie-rod, and in which the masses of said first and said second vibration-damping bodies are substantially equal in magnitude.

3. A handle as claimed in claim 1, wherein said first and said second vibration-damping bodies are each slidably attached on a common tie-rod to permit adjustment of their respective operating positions on the tie-rod.

4. A handle as claimed in any one of claims 1 to 3, in which said hand-grip is tubular and made of a semi-rigid material.

5. A handle as claimed in any preceding claim, in which said tie-rod or each of said tie-rods is made of piano wire.

6. A handle as claimed in claim 2, or as claimed in claim 4 or claim 5 when dependent upon claim 2, in which said second vibration-damping body is slidably attached on the second tie-rod to permit adjustment of its position on the second tie-rod relative to the first vibration-damping body.

7. A handle as claimed in any preceding claim, in which said hand-grip is closed at its end remote from said first vibration-damping body.

8. A handle as claimed in claim 2, or any one of claims 4 to 7 when dependent upon claim 2, in which said first tie-rod is made of a piano wire having a diameter of about 6 mm and said second tie-rod is made of a paino wire having a diameter of about 4.5 cm.

9. A vibration-damping handle substantially as described with reference to figure 6 or figure 7.

10. A power tool having a handle as claimed in any preceding claim.

11. A vehicle having a control hand-grip provided by a handle as claimed in any one of claims 1 to 9.

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